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RESEARCH PAPER

INFLUENCE OF SUB-COOLING ON THE ENERGY PERFORMANCE OF TWO ECO-FRIENDLY R22 ALTERNATIVE REFRIGERANTS

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ABSTRACT
In this study, the effects of sub-cooling on the various refrigeration cycle performance parameters using the alternative refrigerants (R432A and R433A) as working fluids were evaluated theoretically and compared with those obtained using the baseline refrigerant (R22). The results obtained showed that the thermodynamic properties of R432A and R433A matched those of R22 and they could be used as substitute for R22. The two alternative refrigerants exhibited very high thermal conductivity with average values of 59.3% and 58.0% higher than that of R22, respectively. Their condenser duties are also better than that of R22, which shows that they could perform very well as refrigerants in heat pump systems. They exhibited higher coefficient of performance (COP) and higher relative capacity index (RCI) than R22. The average COPs obtained for R432A and R433A were 12.9 and 16.7% higher than that of R22. They also exhibited lower power per ton of refrigeration (PPTR) than that of R22, but R433A emerged as the most energy efficient refrigerant among all the investigated refrigerants with average reduction in PPTR of 28.5% higher than that of R22. Generally, incorporation of sub-cooling heat exchanger in the system, greatly improved the performance of the system; it increased the COP, reduced the compressor energy input and the specific power consumption of the system. The two alternative refrigerants, consistently exhibited better performance than R22 in sub-cooling heat exchanger refrigeration system. R433A performed better than both R22 and R432A in that the highest RCI, COP, reduction in energy input, reduction in PPTR and lowest discharge temperature were obtained using R433A in the system.

Keywords: *Sub-cooling, energy, performance, eco-friendly, R22 alternatives*

INTRODUCTION
Since R22 came into common use as a refrigerant in 1936, it has been applied in systems ranging from the smallest window air-conditioners to the largest chillers and heat pumps because of its inherent efficiency and

high refrigeration capacity and it has the largest sales volume among all refrigerants. Individual equipment using this versatile refrigerant ranges from 2kW to 33 MW in cooling capacity. No other refrigerant has achieved such a wide range of applications (Calm and Domanski, 2004). However, R22 is one of a class of chemicals, HCFCs, to be phased out internationally by year 2020 and 2030 in developed and developing nations, respectively, due to their environmental hazard of ozone depletion (Aprea and Renno, 2004; Bolaji, 2011).

The research on refrigerant replacement for R22 has been one of the hot topics in the refrigeration and air-conditioning industry. No single -component fluid has been identified as a replacement for R22 that would meet all performance, environmental, and safety requirements (Kruse, 2000). Several alternatives, including binary, ternary and quartet blends of hydrofluorocarbons (HFCs) have been considered as potential replacement fluids, since mixing two or more refrigerants can create a new working fluid with the desired characteristics. Nevertheless, the two prominent HFC mixtures (R407C and R410A) developed as alternative to R22 refrigerant have relatively high GWP (Table 1). For this reason, environmentally benign, 'natural' refrigerants have attracted considerable attention.

The natural refrigerants are the naturally occurring substances such as ammonia, hydrocarbons, carbon dioxide, water and air. In this group, the hydrocarbons are most closely related to the HFCs. Their thermodynamic and transport properties are very similar to most HFCs currently used in refrigeration and air-conditioning systems, which make them suitable as substitute refrigerants in the existing HCFC and HFC systems without any major changes in the design (Castro *et al*., 2005; Park and Jung, 2007; Palm, 2007).

Hydrocarbon refrigerants have both zero ODP and very low GWP (Table 1). They are compatible with common materials found in refrigeration and air-conditioning systems and are soluble in conventional mineral oils (Palm,

Table 1: Environmental and thermophysical properties of investigated refrigerants compared with existing refrigerants

Environmental and thermo-physical properties	**Refrigerants**				
	R22	**R432A**	**R433A**	**R407C**	**R410A**
Critical Temperature (^oC)	96.2	97.3	94.4	86.0	71.8/ 4
Normal boiling point, NBP (^oC)	-40.8	-46.6	-44.6	-44.0	-51.0
Temperature glide (^oC)	-	1.0	0.4	5	0.2
Composition	-	R1270 (80%) RE170 (20%)	R1270 (30%) R290 (70%)	R32 (23%) R125 (25%) R134a (52%)	R32 (50%) R125 (50%)
Molar mass (kg/kmol)	86.5	42.8	43.5	86.2	72.6
Ozone Depleting Potential (ODP)	0.05	0	0	0	0
Global warming potential (GWP) (100 years' horizon)	1500	4	4	1525	1725

Sources: Lemmon et al. (2002); Bitzer (2012)

2008). The most important concern regarding the adoption of hydrocarbons as a refrigerant is their flammability. It should be remembered that millions of tonnes of hydrocarbons are used safely every year throughout the world for cooking, heating, powering vehicles and as aerosol propellants. In these industries, procedures and standards have been developed and adopted to ensure the safe use of the product. The same approach is also being followed by the refrigeration industry. Various applications have been developed in handling the flammable and safety problems such as using enhanced compact heat exchangers, optimizing system designs, reducing the charge of systems and establishing rules and regulations for the safety precautions (Fernando *et al*., 2004; Thonon, 2008).

Sub-cooling heat exchanger is a tool that can be used to evaluate the impact of refrigerants on refrigeration system's capacity and performance (Bolaji *et al*., 2010). It is commonly installed in refrigeration systems with the intent of ensuring proper system operation and increasing system performance (ASHRAE, 1998). This tool has been used by some researchers (Domanski and Didion, 1993; Domanski *et al*., 1994; Klein *et al*., 2000; Bolaji *et al.,* 2010) to evaluate some alternative refrigerants to R22 and R12. Therefore, in this paper, the performance of hydrocarbon mixtures (R432A and R433A) with zero ODP and negligible GWP (Table 1) as alternatives to R22 in vapour compression refrigeration system was investigated theoretically employing a sub-cooling heat exchanger refrigeration system. R432A is a near azeotropic mixture composed of 80% propylene (R1270) and 20% dimethylether (RE170) by weight. R433A is also a near azeotropic mixture composed of 30% propylene (R1270) and 70% propane (R290) by weight. The effects of sub-cooling and performance parameters of the system working with alternative refrigerants were evaluated and compared with those of R22.

MATERIALS AND METHODS

Refrigeration system with sub-cooling

Sub-cooling in refrigeration implies cooling the refrigerant in liquid state, at uniform pressure, to a temperature that is less than the saturation temperature, which corresponds to condenser pressure. Schematic diagram of vapour compression refrigeration system with a sub-cooling heat exchanger is shown in Fig. 1. In this system, high temperature liquid from the condenser is sub-cooled in the heat exchanger before entering the expansion device where it is being throttled to the evaporator pressure. The sub-cooling heat exchanger is an indirect liquid -to-vapour heat transfer device where high temperature and pressure liquid refrigerant transfers heat to the low temperature refrigerant vapour leaving the evaporator. The heat exchanger also prevents the carrying-over of liquid refrigerant from the evaporator to the compressor.

The sub-cooling heat exchanger affects the performance of a refrigeration system by influencing both the high and low pressure sides of the system. Fig. 2 shows the key state points for a vapour compression cycle utilizing an idealized sub-cooling heat exchanger on a pressure-enthalpy diagram. Degree of sub-cooling is the difference between the saturation temperature of the liquid refrigerant corresponding to condenser pressure and the temperature of the liquid refrigerant before entering to the expansion device.

Relative capacity index (*RCI*)

Without a sub-cooling heat exchanger, the refrigerating effect per unit mass flow rate of circulating refrigerant is the difference in enthalpy between states 1 and 4 in Fig. 2. When the heat exchanger is installed, the refrigeration effect per unit mass flow rate increases to the difference in enthalpy between states '1' and '44'. If there were no other effects, the addition of a sub-cooling heat exchanger would always lead to an increase in the refrigeration capacity of a system. the extent of the capacity increase is a function of the specific heat of refrigerant,

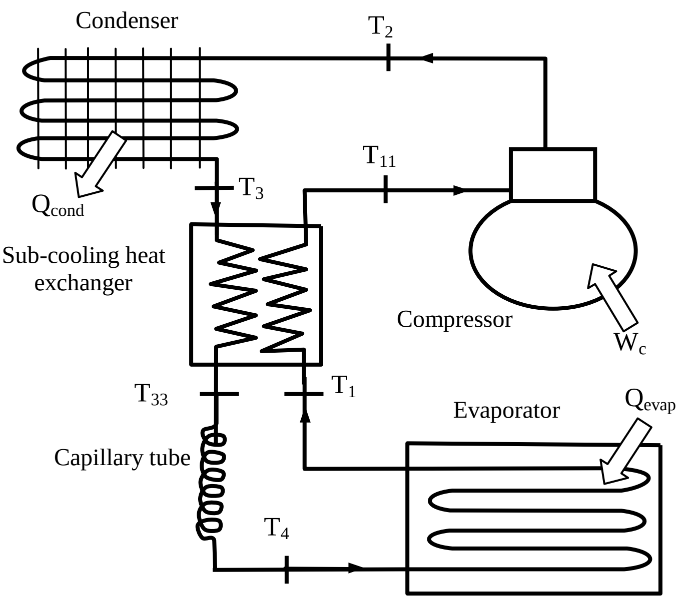

Fig. 1: Refrigeration system with a sub-cooling heat exchanger

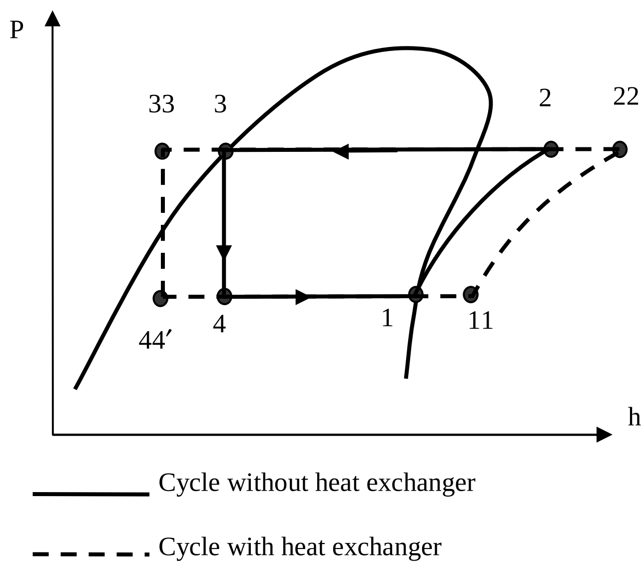

Fig. 2: Pressure-enthalpy diagram showing effect of an idealized sub-cooling

the degree of sub-cooling and the system operating conditions. according to klein and reindl (1998), the effect of a sub-cooling heat exchanger on refrigeration capacity can be quantified in terms of a relative capacity index (*rci*) as defined in eq. (1):

$$RCI = \left(\frac{RC_{hx} - RC_{nohx}}{RC_{nohx}} \right) \times 100\% \quad (1)$$

where, RC_{hx} = the refrigeration capacity with a sub-cooling heat exchanger; and RC_{nohx} = the refrigeration capacity for a system operating at the same condensing and evaporating temperatures without a sub-cooling heat exchanger.

Refrigeration cycle performance calculations were carried-out with the assumption that the refrigerant exits the evaporator as a saturated vapour at the evaporator pressure (state 1) and exits the condenser as a saturated liquid at the condenser pressure (state 3). When a sub-cooling heat exchanger is employed, the refrigerant entering the compressor (state 11) has been superheated by heat exchange with the liquid exiting the condenser, which causes the liquid to enter the expansion device in a sub-cooled state (state 33).

Determination of thermodynamic properties of investigated refrigerants

The most fundamental of a working fluid's thermal properties that are needed for the prediction of a refrigerant system's performance are the pressure-volume-temperature (PvT) in an equilibrium state. Other properties, such as enthalpy and entropy as well as the Helmholtz and Gibbs functions, may be derived from a PvT correlation utilizing specific heat. There exists a myriad of equations-of-state, which have been classified into families. These equations have been used to develop the most widely used refrigerant database software known as REFPROP (Lemmon *et al*., 2002; Didion, 1999). It was developed and is maintained by the National Institute of Standards and Technology and is currently in its ninth edition. It uses several equations-of-state to correlate 33 single component refrigerants and 29 predefined mixtures, along with the ability to construct virtually any desired mixture of up to five components (Kumar and Rajagopal, 2007). This software was used in this work to compute the properties of investigated refrigerants.

Data reduction

After the thermodynamic properties of each state of the cycle are determined, the equations for the cycle analysis are obtained by means of mass and energy conservation. The data reduction of the theoretical results are analysed with the equations stated below. Considering the cycle on p-h diagram in Fig. 2, the heat absorbed by the refrigerant in the evaporator or refrigerating effect (Q_{evap}, kJ/kg) is calculated as:

$$Q_{evap} = (h_1 - h_4) \quad (2)$$

where, h_1 = specific enthalpy of refrigerant at the outlet of evaporator (kJ/kg); and h_4 = specific enthalpy of refrigerant at the inlet of evaporator (kJ/kg). The compressor work input (W_{comp}, kJ/kg) is obtained as:

$$W_{comp} = (h_2 - h_1) \quad (3)$$

where, h_2 = specific enthalpy of refrigerant at the outlet of compressor (kJ/kg). The flow of refrigerant in the throttling valve from point 3 to point 4 is at constant enthalpy (isenthalpy). Therefore,

$$h_3 = h_4 \quad (4)$$

where, h_3 = specific enthalpy of refrigerant at the outlet of condenser (kJ/kg). The specific power consumption is a useful indicator of the energy performance of refrigeration system. This is obtained as Power per ton of refrigeration (*PPTR*) and is expressed as (Dalkilic and Wongwises, 2010):

$$PPTR = \frac{3.5W_{comp}}{Q_{evap}} \quad (5)$$

The coefficient of performance (COP) is the refrigerating effect produced per unit of work required; therefore, COP is obtained as the ratio of Eq. (2) to Eq. (3):

$$COP_{ref} = \frac{Q_{evap}}{W_c} \quad (6)$$

RESULTS AND DISCUSSION

The variation of saturated vapour pressure and temperature for R22 and its alternative refrigerants (R432A and R433A) is shown in Fig. 3. As shown in this figure, the saturated vapour pressure curves for R432A and R433A are almost the same with that of R22 without any significant deviation between the curves. This indicates that R432A and R433A can exhibit similar properties and could be used as substitute for R22.

The variation of thermal conductivity with the degree of sub-cooling for R22 and its two alternative refrigerants at condensing temperature of 40°C and evaporating temperature of -20°C is shown in Fig. 4. Refrigerant thermal conductivity slightly increases as degree of sub-cooling increases. The two alternative refrigerants (R432A and R433A) exhibited higher thermal conductivity with average values of 59.3% and 58.0% higher than that of R22, respectively.

Fig. 5 shows the influence of the degree of sub-cooling on refrigerating effects for R432A, R433A and R22 at condensing temperature of 40°C and evaporating temperature of -20°C. As shown in the figure, refrigerating effect increases with increase in degree of sub-cooling. This is due to the increase in latent heat value of the refrigerant. A very high latent heat value is desirable since the mass flow rate per unit of capacity is less. When the latent value is high, the efficiency and capacity of the compressor are greatly increased. It is clearly shown in Fig.

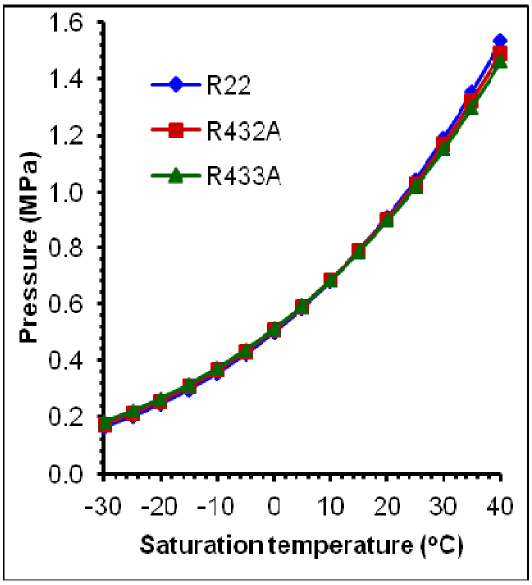

Fig. 3: Saturation vapour pressure curves

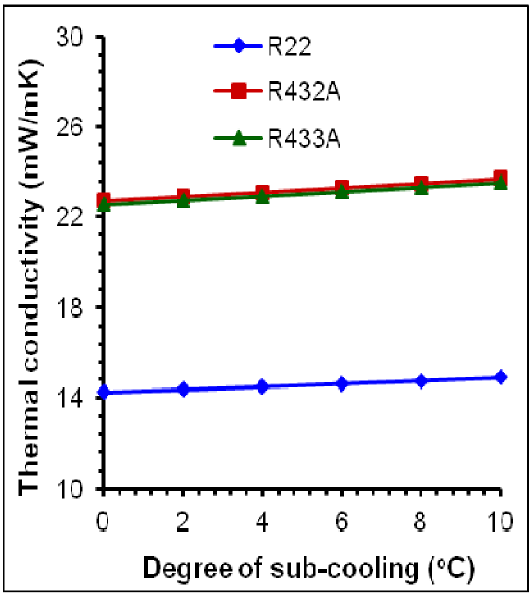

Fig. 4: Variation of thermal conductivity with degree of sub-cooling at 40°C condensing and -20°C evaporating temperatures

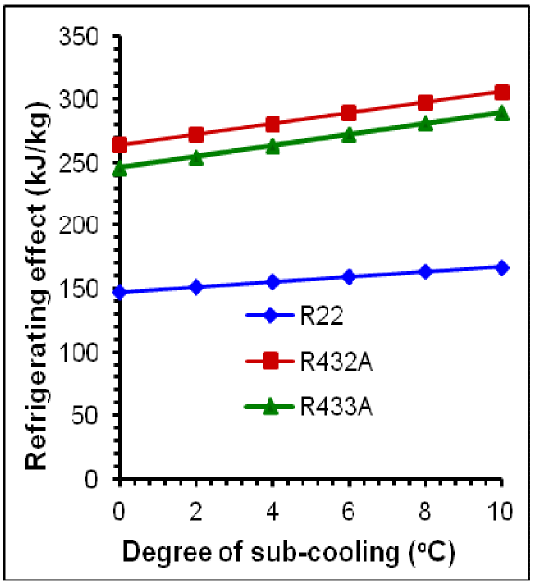

Fig. 5: Influence of the degree of sub-cooling on refrigerating effect at 40°C condensing and -20°C evaporating temperatures

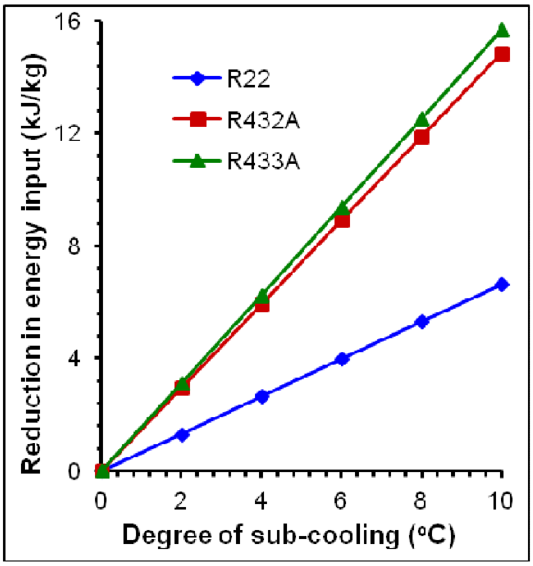

Fig. 6: Effect of the degree of sub-cooling on the reduction in compressor energy input at 40°C condensing and -20°C evaporating temperatures

5 that R432A and R433A exhibited higher refrigerating effect than R22. Therefore, very low mass of refrigerant will be required for the same capacity, and smaller compressor size will also be required due to their high latent heat values. The highest refrigerating effect was obtained using R432A in the system.

Fig. 6 shows the reduction in the compressor energy input for the investigated refrigerants at varying degree of sub-cooling for 40°C condensing and -20°C evaporating temperatures. This figure revealed that reduction of energy input increases with increase in degree of sub-cooling. The two alternative refrigerants exhibited higher reduction in energy input than R22, but the highest reduction in energy was obtained using R433A in the system.

The effect of the degree of sub-cooling on the discharge temperature at 40°C condensing and -20°C evaporating temperatures for the three investigated refrigerants is shown in Fig. 7. As shown in the figure, the alternative refrigerants (R432A and R433A) exhibited lower values of discharge temperature than R22. This is an added advantage for the two alternatives. High discharge temperature is detrimental to the performance of the system, therefore, low discharge temperature is required, which means that there will be less strain on the compressor and hence a longer compressor life. The average discharge temperature obtained for R432A and R433A were 18.3% and 26.9% lower than that of R22, respectively.

The influence of the degree of sub-cooling on the condenser duty at 40°C condensing and -20°C evaporating temperatures for the investigating refrigerants is shown in Fig. 8. As shown in the figure, the condenser duty increases as degree of sub-cooling increases. The increase in the degree of sub-cooling reduces the temperature of the liquid refrigerant at the exit of the condenser and therefore increases the quantity of heat to be removed by the condenser. It is clearly shown in Fig. 8 that the condenser duty using R432A and R433A are better than

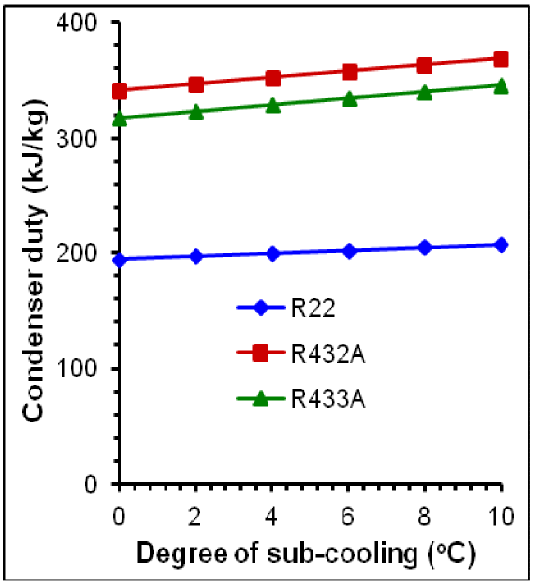

Fig. 8: Influence of the degree of sub-cooling on condenser duty at 40°C condensing and -20°C evaporating temperatures

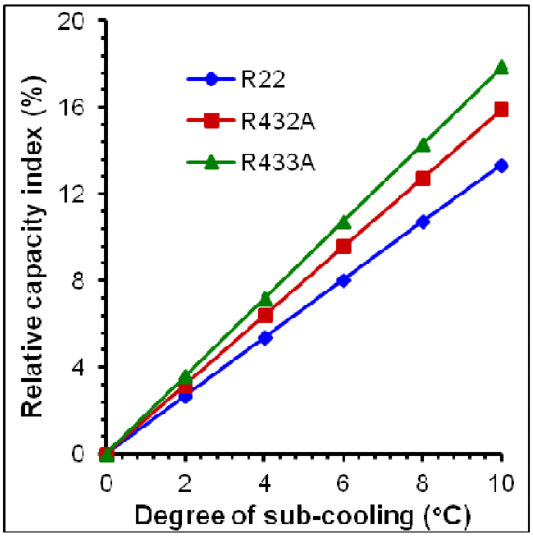

Fig. 9: Effect of the degree of sub-cooling on the relative capacity index (RCI) at 40°C condensing and -20°C evaporating temperatures

that of R22. This shows that they could perform very well as refrigerants in heat pump systems. The highest condenser duty was obtained using R432A.

Fig. 9 shows the effect of the degree of sub-cooling on the relative capacity index (RCI) for R22, R432A and R433A at 40°C condensing and -20°C evaporating temperatures. As shown in this figure, an increase in capacity is observed for all refrigerants, although there is considerable variation in the magnitude of the effect of sub-cooling on each refrigerant. The average relative capacity indices obtained for R432A and R433A were 19.2% and 33.6% higher than that of R22.

The coefficient of performance (COP) of refrigeration cycle reflects the cycle performance and is the major criterion for selecting a new refrigerant as a substitute. Fig. 10 shows the effect of degree of sub-cooling on the COP at 40°C condensing and -20°C evaporating temperatures for R22 and the two alternative refrigerants. This figure clearly shows the effect of sub-cooling on the refrigerant performance. At zero degree sub-cooling (without sub-cooling), the COP of R22 was very close to those of R432A and R433A, but the degree of sub-cooling increased the COP of the two alternative refrigerants more than that of R22. The average COPs obtained for R432A and R433A were 12.9% and 16.7% higher than that of R22.

Fig. 11 shows the influence of the degree of sub -cooling on the power consumption per ton of refrigeration at 40°C condensing and -20°C evaporating temperatures. As shown in the figure, the power per ton of refrigeration (PPTR) reduces as the degree of sub-cooling increases for all the investigating refrigerants. R432A and R433A exhibited lower PPTR with the average values of 10.9 and 13.3% lower than that of R22, respectively. Also, Fig. 12 shows the reduction in PPTR for the investigated refrigerants at varying degrees of sub-cooling. This figure revealed that reduction in power consumption increases as the degree of sub-

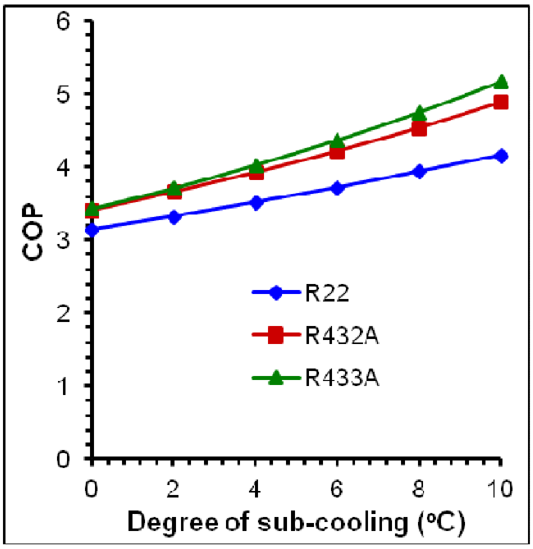

Fig. 10: Effect of the degree of sub-cooling on the coefficient of performance (COP) at 40°C condensing and -20°C evaporating temperatures

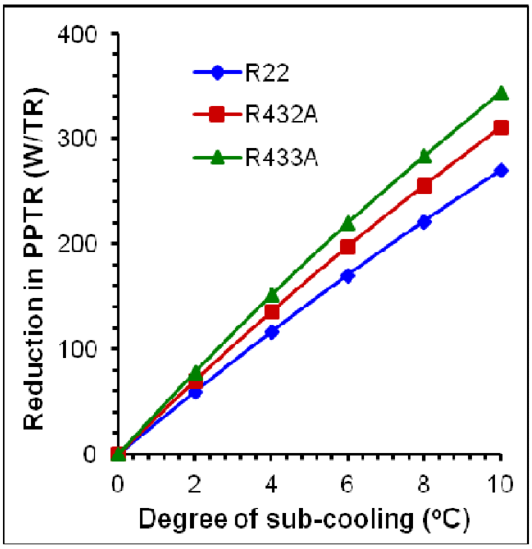

Fig. 12: Effect of the degree of sub-cooling on the reduction in power per ton of refrigeration (PPTR) at 40°C condensing and -20°C evaporating temperatures

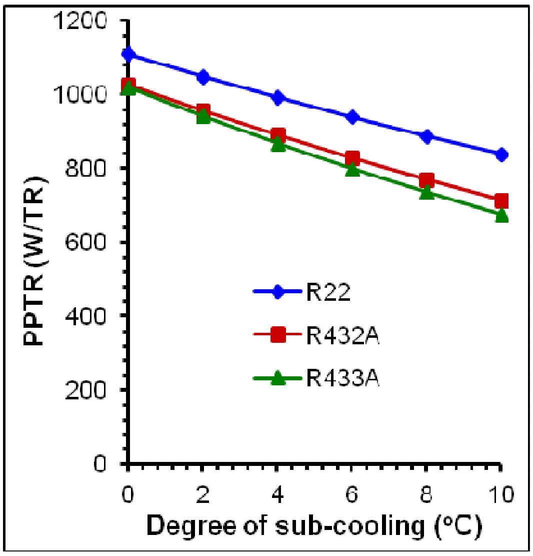

Fig. 11: Influence of the degree of sub-cooling on the power per ton of refrigeration (PPTR) at 40°C condensing and -20°C evaporating temperatures

cooling increases. R432A and R433A refrigerants exhibited higher reduction in power than R22, with average values of 15.7 and 28.5 higher than that of R22, respectively.

CONCLUSION

In this study, the energy performance of eco-friendly R432A and R433A as alternatives to R22 in vapour compression refrigeration system was investigated theoretically employing a sub-cooling heat exchanger refrigeration system. The effects of sub-cooling on the various refrigeration cycle performance parameters were evaluated. Based on the investigation results, the following conclusions are drawn:

(i) The saturated vapour pressure and temperature characteristic profiles for R432A and R433A are very close without any significant deviation between the curves. This indicates that R432A and R433A exhibited similar properties and could be used as substitute for

R22

(ii) R432A and R433A exhibited very high thermal conductivity with average values of 59.3% and 58.0% higher than that of R22, respectively.

(iii) R432A and R433A exhibited higher refrigerating effect than R22. Therefore, very low mass of refrigerant will be required for the same capacity, and smaller compressor size will also be required due to their high latent heat values.

(iv) The reduction in the compressor energy input using the two alternative refrigerants is higher than that of R22, but the highest reduction in energy was obtained using R433A in the system.

(v) The two alternative refrigerants exhibited lower values of discharge temperature than R22. The average discharge temperature obtained for R432A and R433A were 18.3% and 26.9% lower than that of R22, respectively.

(vi) The condenser duty using R432A and R433A are better than that of R22. The highest was obtained using R433A. The high condenser duty of the alternative refrigerants shows that they could perform very well as refrigerants in heat pump systems.

(vii) The effect degree of sub-cooling has on the refrigerating capacity is higher for the two alternative refrigerants than for R22. The highest relative capacity index (RCI) was obtained using R433A in the system.

(viii) The two alternative refrigerants exhibited higher coefficient of performance (COP) than R22. The average COPs obtained for R432A and R433A were 12.9% and 16.7% higher than that of R22.

(ix) The two alternative refrigerants exhibited lower power per ton of refrigeration (PPTR) than that of R22, but R433A emerged as the most energy efficient refrigerant among all the investigated refrigerants with average reduction in PPTR of 28.5% higher than that of R22.

Generally, incorporation of sub-cooling heat exchanger in the system, greatly improved the performance of the system; it increases the COP, reduces the compressor energy input and the specific power consumption of the system. The thermodynamic properties of R432A and R433A matched those of R22 and the two refrigerants consistently exhibited better performance than R22 in sub-cooling heat exchanger refrigeration system. R433A performed better than both R22 and R432A in that the highest RCI, COP, reduction in energy input, reduction in PPTR and lowest discharge temperature were obtained using R433A in the system.

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